11 The Power Station

The prime mover for bulk electricity generation worldwide is the turbine, coupled to a condenser and alternator. A working fluid is taken through a cycle of changes of temperature and pressure and through the prime mover to produce torque, which involves heat supply and rejection to the surrounding environment. The dominant choice of working fluid remains as water, and the steam cycle has undergone over a century of development. Geothermal steam is not supplied as part of a closed cycle, but turbines for conventional (fossil-fuelled) power stations have required only relatively minor modifications in design to make use of it. However, the low temperature of most geothermal heat supplies compared to fossil-fuelled boilers has led to the choice of organic fluids as the working fluid, for which the design of the power station equipment, including the turbine and condenser, must be altered quite significantly.

This chapter discusses historical trends in steam cycle power stations and then explains the development of the steam Rankine cycle and the modifications introduced to make it approach the Carnot cycle more closely. As well as providing the background for geothermal steam plant, it provides a starting point for considering designs for alternative working fluids. Details of turbines, condensers and ancillary equipment for geothermal steam power stations are introduced and organic Rankine cycle and trilateral flash cycle plant are discussed.

11.1 Introduction

Electricity is generated by the rotation of a coil in a magnetic field and the power station houses the machinery to do this. Since about 1900 most power stations have used steam turbines as the drivers (prime movers) and fossil fuels as the heat source. The use of geothermal steam to drive a turbine is a departure from fossil-fuelled practice in that the steam enters the station from the wells and leaves it as condensate—the turbine is a machine placed in a once-through steam flow. In fossil-fuelled stations a boiler generates the steam which passes through the turbine into a condenser and back to the boiler. This is convenient because the water can be

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maintained at a high level of purity, reducing corrosion so as to allow metals to be used at high temperature over a long working life. But it is not merely a convenience; it complies with the concept of a heat engine as originated by Carnot. A heat engine is defined as a machine which takes a fixed mass of fluid—the working fluid—through a cycle of processes during which heat is transferred to and from the fluid and mechanical work is produced. Thus a steam turbine using geothermal steam is not strictly a heat engine, as it does not operate continuously on the same mass of steam. This is a theoretical rather than a practical matter, and geothermal steam turbines are virtually identical to those used in fossil-fuelled stations, except for design differences necessitated by the relatively low steam temperature and pressure used compared with modern fossil-fuelled plant. The turbine is essentially an item of machinery within which the momentum of the steam flow is transferred to the rotating shaft; it is not integral to the steam cycle and in fact very little of turbine blade design involves thermodynamics. The same issue regarding the cycle arises in internal combustion engines, which continuously refresh the working fluid, yet their performance is calculated as if they were true heat engines.

Despite this, there is merit in studying the Rankine cycle as applied to fossil-fuelled generation for consideration of geothermal turbines. It is the main cycle for steam plant using water as working fluid and has direct application to geothermal turbines using working fluids other than water—organic fluids—which must be conserved for cost and environmental reasons, so are used in a true heat engine cycle. The Carnot cycle is the ideal, regardless of working fluid, and the Rankine cycle departs from it but is the closest practical alternative. Various modifications to the cycle have been adopted over the last century to bring it closer to the ideal, and these have led to modifications to plant design. The fossil-fuelled steam Rankine cycle is a datum, and once understood, the variations introduced for geothermal application follow very easily. It will be found that the design of the power plant involves a good deal of optimisation. In common with most heavy engineering equipment, the actual design details are commercially sensitive, and a purchaser must understand the general operation of the equipment at a fundamental level (or engage a consultant) for a critical examination of it.

It is not proposed to deal with electricity per se in this book. As noted, Faraday established the principle of generation in 1831, but a generator was not built until the 1870s; a brief history is given by van Riemsdijk and Brown [1980]. For large-scale generation, three-phase alternating current has been usual since the early days, and the generator is called an alternator. The voltage frequency is either 50 or 60 Hz, produced by rotating the alternator at 3,000 or 3,600 rpm, respectively. The alternator is driven by a prime mover, a steam turbine for the largest electrical output, and the turbine is usually designed to run at the same speed as the alternator. Turbine and alternator are usually coupled directly, with their axes in line, the pair being known as a turbo-generator or more specifically a turbo-alternator. Turbines are designed to rotate at "synchronous speed", i.e. to give 50 or 60 Hz electrical output, and precise speed control is essential. Gear boxes to alter speeds are sometimes used but only for smaller power outputs, since gearboxes are expensive and involve greater wear and tear than turbines or alternators; turbines using

organic working fluids do not suit such high-speed rotation as steam turbines and may use gearboxes. A transmission grid is required to distribute the electricity produced, usually operating at 440 kV or more, much higher than the generation voltage of typically 11 kV, and turbo-alternators feed into transformers and a switchyard for connection to the grid (see Fig. 1.2). There are a few electrical equipment design issues specific to geothermal power stations. One of these is the abnormally high concentration of H_2S in the atmosphere around geothermal power stations, due partly to the natural surface activity to be found in many geothermal resource areas but partly also to the discharge to atmosphere of the non-condensable gas extracted from the condenser. The gas causes corrosion of the copper used in exposed switches and related conductors. Another issue is the earthing mat, an extensive mesh of copper conductors buried beneath the power station foundations to provide a good earth connection as a datum voltage level. Geothermally altered soils provide a very acidic, corrosive environment for buried copper conductors.

11.2 Historical Trends in Thermal Power Stations

The dominant feature in distributed electricity supply of all types, not just geothermal, over the 120 years of its existence has been the growth in consumption and hence installed capacity, and this is expected to continue. It has been estimated by the IAEA [2012] that world electricity supply will double in the period 2000–2030, requiring an investment equivalent to US$550 billion per annum for new and replacement plant. Approximately half of the investment goes to generation and half to distribution equipment. Steam has been and will continue to be the dominant working fluid used to drive turbo-alternators, typical single machine outputs of which have increased from 1 MWe in 1900 to 25 MWe in 1912, 128 MWe in 1928, 600 MWe in 1960, and so on, with the largest single unit being in the order of 1,000 MWe at the present time.

Efficiency has been an important consideration throughout. In fact it has been an important consideration since the first days of steam engines—those used for draining British coal mines burned so much coal that where possible the water was discharged upstream of a watermill to recover some of the value. Improvements in efficiency over the period to about 1970 were made as the result of continuous increase in steam temperatures and pressures; by the 1960s steam temperatures had reached 550 °C or thereabouts, a level not much exceeded because of blade material problems (creep). Geothermal steam conditions were far below those of contemporary fossil-fuelled stations when geothermal resources came into use in the period 1950–1960 and represented a step backwards in the efficiency of conversion of heat to electricity; the same was true of nuclear-generated steam at that time.

Pressures in modern fossil-fuelled steam plant are now often supercritical. The "oil shock" of the 1970s stimulated further research and development; there was concern about the supply of petroleum fuels worldwide and a consequent increase in their price. Combined cycle plant became common, gas turbines with their

exhaust providing the heating to boilers now termed steam generators, with thermal efficiencies well above 40 %. Attention was also paid to working fluids other than water in the search for ways to make better use of lower temperature heat sources. Alternative working fluids had been considered much earlier, mercury in 1923 and diphenyl-oxide in 1934. Both of these have a very high critical temperature and moderate critical pressure, advantageous for use in the high-temperature part of the cycle, whereas organic fluids were aimed at application in the low-temperature part of the cycle, as waste heat recovery units. Small unit sizes of the latter were developed, for example, by Ormat Ltd, which could stand alone to recover heat from suitable sources, and the general type became known as organic Rankine cycle (ORC) plants. This began a trend to smaller unit sizes, which suited application to geothermal energy, and a departure from the practice of housing all generating plant in a building. Free-standing units exposed to the weather became possible, and the task of bringing into use resources in remote locations was made easier and more economic. The ultimate capacity of a newly explored geothermal resource and the capital cost of the exploration make it appropriate to build power stations in stages, which the availability of smaller units also helps.

In the present era, global warming and general concern for the environment are additional driving forces for the development of geothermal generation and use of low-temperature fluids, and a second round of the search for alternative working fluids is evident.

11.3 Generation of Electricity Using Steam as the Working Fluid in a Cycle

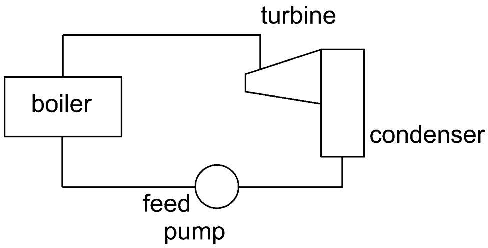
11.3.1 The Rankine Cycle as a Representation of the Carnot Cycle

The Carnot cycle was introduced in Chap. 3 as a means of defining entropy; a fixed mass of fluid was taken through four processes which changed its thermodynamic state but brought it back to the starting point. The fluid did not flow; it had no velocity at any stage but merely changed its thermodynamic state as it was presented to four different thermal boundary conditions in turn. Now the focus is on the steady flow of H_2O around the circuit shown in Fig. 11.1, a boiler supplying steam to a turbine and condenser, with the liquid water pumped back to the boiler.

A large mass of fluid flows continuously around a circuit, passing through items of equipment that change its thermodynamic state. Heat is added to the loop and work is extracted but all rates of change are zero—any parameter measured at any fixed point in the loop is invariant with time. In thermodynamic terms this is described as a closed cycle plant. The fluid now has velocity, and every particle of it passes through a cycle just as did the fixed mass in Chap. 3.

The Carnot cycle is the ideal and can only be approximated in practice. Figure 11.2 shows both the Carnot cycle, GCDFG, and the Rankine cycle,

Fig. 11.1 Fossil-fuelled power station flow path



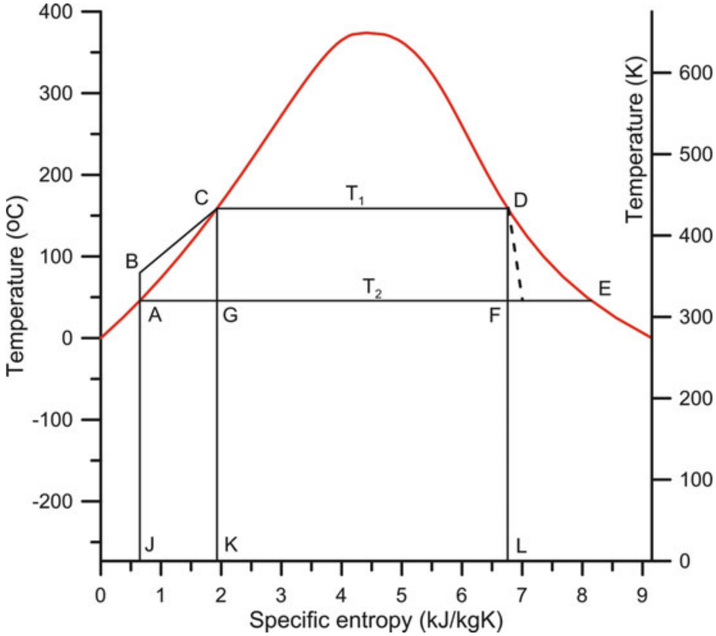


Fig. 11.2 The Carnot and Rankine cycles on a T–s chart for H_2O

ABCDFA, on a T–s diagram for water. The envelope of saturation conditions can be plotted from the steam tables; for other fluids it has a different shape which has significant implications to be discussed later. It will be recalled that the horizontal lines across the envelope represent the path taken by fluid changing from saturated water at C to saturated steam at D, and in reverse from E to A. The fluid is two-phase everywhere inside the envelope. The Carnot cycle has two isothermals, CD at temperature T_1 and FG at T_2, and two isentropic stages, GC and DF. It was not conceived with water and steam in mind as the working fluid, but with a single-phase gas. An isothermal supply and rejection of heat to/from the working fluid are easily achieved because condensation and evaporation are isothermal processes. However, halting the condensation from F exactly at G would be difficult to arrange economically, even today, and the isentropic expansion and compression processes are theoretical ideals. Rankine presented an achievable cycle in 1854 as the best

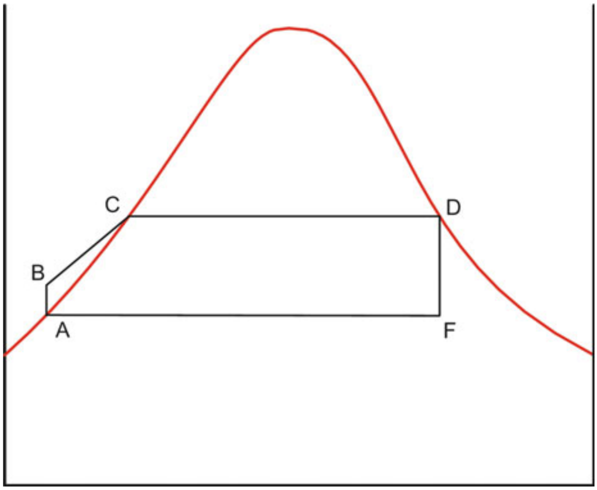


Fig. 11.3 The Rankine cycle on a T–s chart for H_2O

that can be done to approximate the Carnot cycle, and it is named after him. It applies to both reciprocating engines and turbines, but a turbine will be assumed here (Fig. 11.3).

Choosing point A as the beginning of the Rankine cycle, where the working fluid is saturated liquid at the lower temperature, the liquid pressure is raised until it reaches the boiler pressure at B, after which it is heated in the boiler at constant pressure, following the line BC. It begins boiling at C and at low to moderate upper temperatures, T_1, the evaporative heat addition over CD is much greater than the heat addition over BC, so the cycle approximates the Carnot cycle by receiving most of its heat at constant temperature. Work is done on the liquid over path AB, but since water is very incompressible, this is small compared to the work extraction when the saturated steam at D is expanded through the turbine, path DF—steam is very compressible. The expansion DF is not truly isentropic due to friction in the turbine flow passages—the real path is shown dotted in Fig. 11.2; the friction converts mechanical work into an extra source of heat, thus departing from Carnot's ideal circumstances. At the end of expansion, F, the fluid enters the condenser where it is completely condensed to saturated liquid at A. Heat is removed isothermally over path FA, and the process departs from the Carnot cycle only in not stopping at G.

The parameters at various points around the cycle can be calculated in an "informal" way because the mass flow rate is constant throughout—only the steady flow energy equation in its simplest form need be used, together with the definitions of specific entropy, specific enthalpy and dryness fraction. By informal is meant that though the equations of continuity of mass and energy are being solved at each point, it is not necessary to write both down formally. For the Carnot cycle, suppose the upper temperature in Fig. 11.2 is the saturation temperature for a pressure of 6 bars abs, which is 158.8 °C, and the condenser operates at 0.1 bar abs for

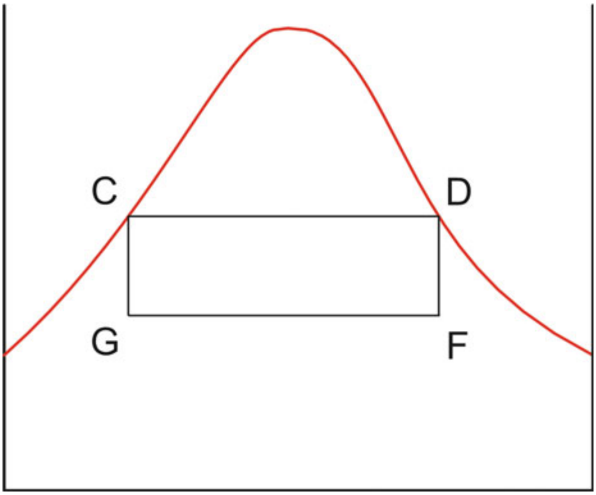


Fig. 11.4 The Carnot cycle on a T–s chart for H_2O

which the saturation temperature is 45.8 °C. The properties required for the calculation are as follows:

Ps (bar abs)	T_s (°C)	S_f	S_{fg}	S_g	h_f	h_{fg}	h_g
6.0	158.83	1.931	4.828	6.759	670.5	2085.6	2756.1
0.1	45.81	0.649	7.500	8.149	191.8	2392.1	2583.9

The units of specific entropy s are kJ/kgK and of specific enthalpy are kJ/kg.
With Fig. 11.4 as reference, first find X_G, the dryness fraction at G:

$$1.931 = 0.649 + X_G^* 7.500 \text{kJ/kgK}$$

giving $X_G = 0.1709$ and thus $h_G = 191.8 + 0.1709 * 2392.1 = 600.69$ kJ/kg
Now find X_F:

$$6.759 = 0.649 + X_F^* 7.500 \text{ kJ/kgK}$$

giving $X_F = 0.8147$ and thus $h_F = 191.8 + 0.8147 * 2392.1$ kJ/kg $= 2140.56$kJ/kg
Using the steady flow energy equation, Eq. (3.6)

$$Q - W = (h_2 - h_1) + \frac{1}{2}\left(u_2^2 - u_1^2\right) + g(z_2 - z_1) \tag{3.6}$$

neglecting the kinetic and potential energy terms and assuming no heat loss, the work output from the turbine and work input in raising the pressure back from condenser pressure to 6 bars abs are, respectively,

$$h_D - h_F = 2756.1 - 2140.56 = 615.54 \text{ kJ/kg}$$

$$\text{and } h_C - h_G = 670.5 - 600.69 = 69.81 \text{ kJ/kg}$$

Thus the net work output from the cycle is 615.54–69.81 = 545.73 kJ/kg.

Alternatively, the net work output can be calculated as the difference between the heat added and the heat rejected, which on Fig. 11.2 is area KCDL—area KGFL which is

$$(158.83 - 45.81)^* s_{fg6\ bar\ abs} = (158.83 - 45.81)^* 4.828 = 545.66 \text{ kJ/kg}$$

These two differ only by rounding errors.

The efficiency of the Carnot cycle, η_C, is the net work output divided by the heat supplied, which is

$$\begin{aligned} \eta_C &= \text{net work output}/T_{S6\ bar\ abs}{}^* s_{fg6\ bar\ abs} \\ &= 545.73/\Big((158.83 + 273.15)^* 4.828 \\ &= 0.2617 \text{ or } 26.17\%. \end{aligned}$$

Alternatively by definition

$$\eta_C = (T_1 - T_2)/T_1 = (158.83 - 45.81)/(158.83 + 273.15) = 0.2616$$

Two arithmetic points should be made here, the dryness fraction should be calculated to four significant figures for steam table work, and the arithmetic was simplified by calculating the temperature difference in °C rather than convert both to degrees K, a dangerous shortcut practice best avoided.

The calculation of efficiency amounts to calculating the ratio of heat supplied to heat rejected in the condenser and is the same for any cycle. It is very simple for the Carnot cycle but less so for the Rankine. The heat rejected is the area of the rectangle JAFL and presents no problem but the heat supplied is the area JABCDL. If the feed pump work is neglected, and the route is assumed to be JACDL, this involves finding the area between the envelope and the temperature axis such as the area shown shaded in Fig. 11.5. This is the Gibbs function for saturated water, which used to be listed in early steam tables for the benefit of steam plant designers.

Since the focus here is on geothermal power plant, full Rankine cycle calculations are not needed, but there are three cycle modifications of significance. The first is the use of feed heating to bring the cycle shape closer to the Carnot cycle; from Fig. 11.2 it can be seen that the main departure from it is the result of being unable to stop the condensation process at G. The significance of a departure can be judged by the change in area from that of the Carnot cycle which it causes, and the change here is JBCK, which is certainly significant compared to KCDL. It represents additional heat discharged to the condenser, which must be made physically larger and more expensive as a result. This departure far outweighs the area modification due to the departure from isentropic of the expansion in

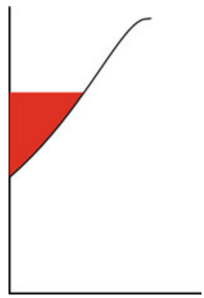


Fig. 11.5 The *shaded area* is the Gibbs function for saturated water

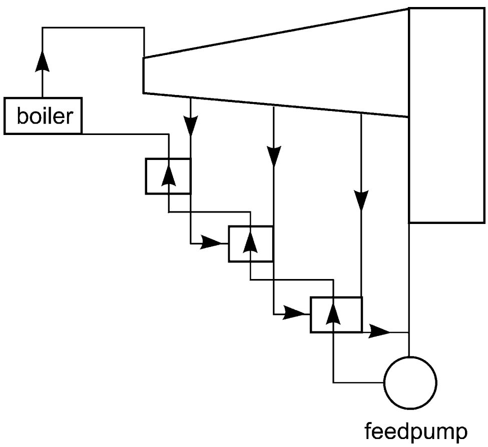


Fig. 11.6 Three stages of feed heating

the turbine. The heating of the condensate takes place at constant pressure, and it was realised early in the twentieth century that taking a small amount of steam from the turbine at various stages through the expansion would provide heat which could be transferred to the condensate, thus bypassing the condenser. Imagine that steam at a temperature halfway between T_1 and T_2 in the expansion on Fig. 11.2 was transferred to the condensate at almost the same temperature. This is possible because the condensation of steam provides heat transfer with only a small temperature difference. The result is a loss of power in the turbine, but a smaller rejection of heat in the condenser, and a consequent saving in condenser cost. This practice is known as regenerative feed heating, and in large fossil-fuelled plant, it is carried out by extracting steam at typically three pressures (temperatures) along the expansion. As shown in Fig. 11.6, the condensate of the highest temperature steam extraction after it has given up its h_{fg} in the first heater is flashed into the second to add to the next steam extraction, and so on. The optimum number of stages is the result of an engineering–economic analysis. When the idea of feed heating was first thought of,

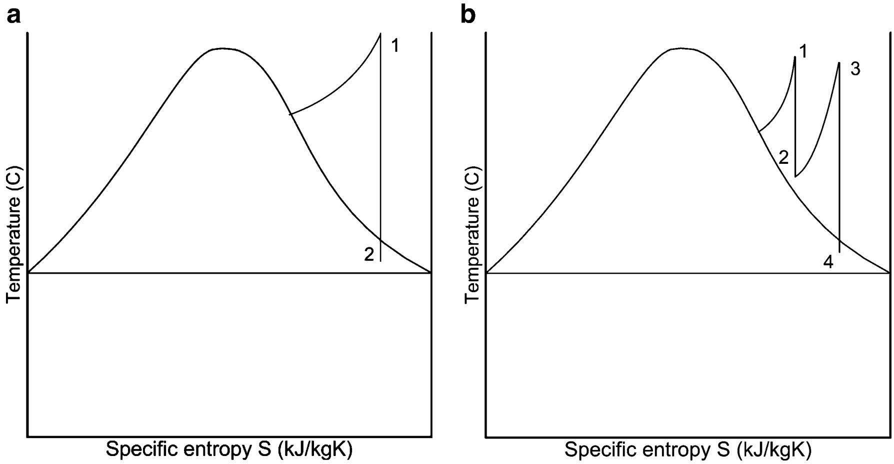


Fig. 11.7 (**a**) Superheating, with expansion from 1 to 2 and (**b**) reheating, with expansion from 1 to 2, then reheating to 3 for expansion to 4

low-temperature boiler exhaust was used to provide the heat, but economic optimisation led to the use of extracted (bled) steam.

The second and third departures from the Rankine cycle involve superheating the steam; what has been referred to as the Rankine cycle so far might be more accurately called the saturated steam Rankine cycle. The saturation envelope has a temperature maximum at the critical point, and the heat added during the evaporation part of the saturated steam cycle becomes a smaller proportion of the total heat added in the cycle as the top temperature T_1 is increased, that is, with increase in boiler pressure. The departure from the Carnot cycle is then greater. Examination of the steam tables reveals that h_{fg} decreases with increasing saturation pressure. The bottom temperature is nominally fixed by atmospheric conditions which represent the lowest practicable heat rejection temperature of perhaps 25–30 °C. Increasingly higher top temperatures were sought to increase the thermal efficiency; however, saturated steam expanded from high pressure becomes very wet in the turbine before it has reached the condenser; water droplets damage the turbine blades by eroding them if the wetness is greater than about 10 %. Figure 11.7a shows one remedy, referred to as superheating, in which the saturated steam emerging from the evaporation part of the boiler is superheated to point 1 by being exposed to higher temperatures in the boiler. This point is at higher entropy than for the original saturated steam and the shape of the saturation envelope is such that moving point 1 to the right makes the steam less wet after it expanded to condenser temperature at point 2.

Figure 11.7b shows the cycle modification referred to as reheat. The steam supply to the turbine is superheated to point 1 and expanded to a temperature in the region of the saturation envelope, point 2, at which it is extracted from the

turbine and passed back through the boiler to be reheated at a lower pressure to point 3 and expanded to condenser conditions at point 4. In practice the turbine can be made up of two separate machines, coupled on the same axis with the alternator, one passing only the high-pressure steam and the other the reheated steam. Optimising the maximum superheat temperature and the end of the first expansion is again a complex engineering—economic problem—and in the context of this book is a problem that would need to be re-examined for every different working fluid.

11.3.2 Steam Turbines in Practice

High-speed reciprocating steam engines were developed in the early part of the twentieth century but were outstripped by turbines. Apart from a device ascribed to Hero of Alexandria in about 50 A.D. which is often used as a way of explaining how a reaction turbine works, the first description of a steam turbine was that by Giovanni Branca (1629) in Italy. Turbines were built and developed in Britain by Parsons (1884), in France by de Laval (1889) and Rateau (1898), in the USA by Curtis (1896) and in Sweden by Ljungstrom (1910)—many were working on the idea at the same time.

There is nothing in the Rankine cycle itself which implies benefit from using a turbine. van Riemsdijk and Brown [1980] quote from a lecture given by Parsons explaining his reasoning, which was that since water turbines have a high mechanical efficiency, 70–80 %, which he put down to water having a small compressibility, it should be possible to make a steam turbine with so many stages that the pressure drop across each would be small enough for the expansion of the steam in that stage to be small also. By analogy, it would be reasonable to expect the same high stage efficiency as for a water turbine, and the mechanical efficiency of the whole multistaged turbine should thus be much higher than that of a reciprocating steam engine, although the thermodynamic efficiency would still be governed by the second law of thermodynamics via the Rankine cycle. It was the mechanical efficiency of the machinery which Parsons was considering. The turbine can be almost perfectly balanced and extracts energy from flow right up to entry to the condenser, whereas in a reciprocating engine, the moving mass of the piston changes direction at the end of each stroke and uses some of the expansion of the steam to clear the cylinder. The turbine was thus ideal for electricity generation, mechanically efficient, requiring low maintenance and having a high rotational speed.

Pictures of turbines are available on the websites of turbine manufacturers such as Mitsubishi Heavy Industries [2012] and others. Large machines might have different turbines mounted on the same shaft, so that the whole expansion of the steam takes place in separate stages of machinery. The separation into stages may be associated with the cycle or for mechanical reasons. Some of the important aspects of turbine design can be illustrated by considering different types of machine.

A condensing turbine exhausts the steam at the lowest practicable temperature for performing work, which will be the temperature for which the condenser was

designed. A back-pressure turbine exhausts steam at a pressure that allows more work or heat to be extracted. The reason may be to allow the steam to be used for some other purpose, such as a chemical processing plant, or to pass the steam back into the boiler for superheating or reheating. A high inlet pressure machine may be split into stages simply for mechanical design reasons, such as a benefit from changing the mean diameter at which the blades rotate. Alternatively, there may be two sources of steam available, the high-pressure steam at inlet and a lower-pressure supply. In this case the turbine exhaust steam and the other supply will be added together, requiring a turbine with a bigger cross-sectional area for flow.

The passage of the steam through the turbine blades produces a rotation which drives the alternator but also produces an axial thrust, which on a large turbine may require a very large bearing surface to carry it. Sometimes turbines are designed to be back to back so the steam flow enters at the centre and divides to flow in opposite directions. The blades must be designed differently, to produce the same rotation, but the axial thrust is cancelled—again an optimisation study is required by the manufacturer.

Figure 11.8 shows a mixed pressure turbine for a nuclear power station, chosen to illustrate the points above. The high-pressure steam enters at the top on the drawing and moves to the left through the high-pressure turbine, which has four stages, each a row of blades preceded by a row of fixed blades known as nozzles to redirect the steam flow. On leaving the high-pressure turbine, the flow direction is changed through 180°, lower-pressure steam is added and the total flow passes through a second turbine with four stages before leaving to be reheated. The steam passage through the second turbine is at a greater radius, providing a bigger area of cross section for the flow, which has increased in mass flow rate by the extra steam added. In accordance with Parsons' ideas, the pressure drop over each stage is kept small. The axial thrust of each of the eight stages is balanced as much as possible, to keep the net axial thrust small; thrust bearings can be incorporated to resist the net thrust but invoke frictional torque and a power loss. The turbine rotational speed is dictated by the alternator output frequency, but several variables remain to manipulate and minimise the net axial thrust.

It can be seen from the figure that the steam can flow through the blades as intended but it can also bypass the blade because there must be a gap between the rotating parts (the shaft and blades) and the stationary parts (the nozzles and their attachment to the fixed outer casing). This bypass route is made as small as possible, as any steam flowing through it is wasted, and this is the reason for the complicated structure in this area. Elaborate sealing methods have been developed.

All axial flow steam turbines look superficially the same. At the high-pressure end, turbine blades are short relative to their diameter, but as the condenser is approached, they must become longer because of the increased cross-sectional area required by the increasing volumetric flow rate, and their active length is at a larger radius. The cross-sectional shape of the passages relative to the turbine axis is indicated by Fig. 11.9 in which the first nozzle is drawn as an actual nozzle, which they sometimes were in older, smaller designs, and the other sets of nozzles as fixed blades.

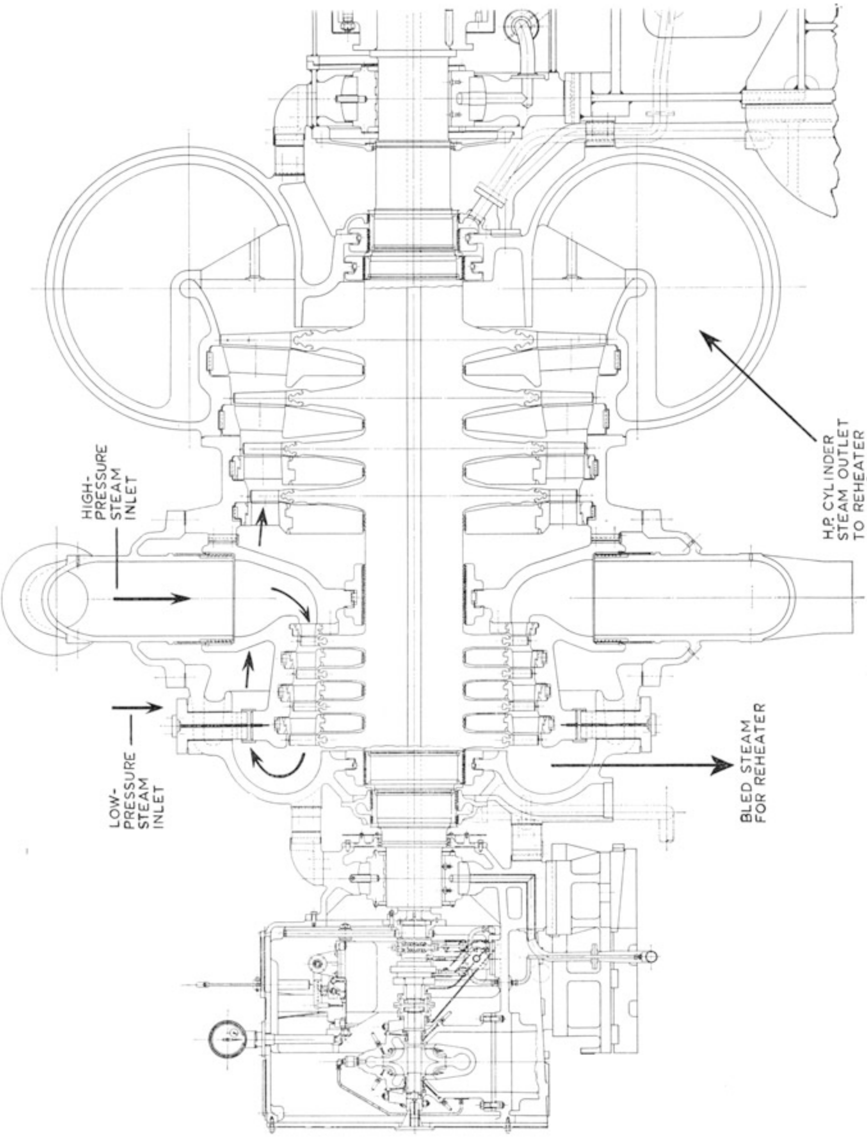


Fig. 11.8 Drawing of part of a 325 MWe mixed pressure turbine illustrating the steam flow path, steam addition after the high-pressure stages and a change in flow direction and passage diameter (reproduced from Worley, Proc (1963–64) by permission of SAGE Publications Ltd.)

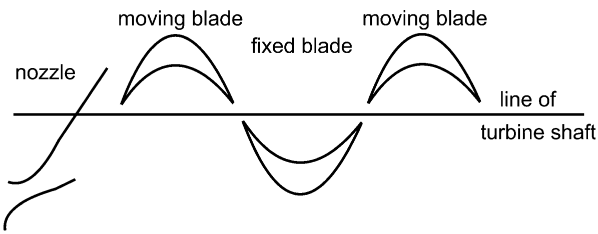


Fig. 11.9 Looking radially inwards towards the turbine axis of rotation, showing a single nozzle and moving and fixed blades. The fixed blades are also referred to as nozzles. The flow passes from left to right

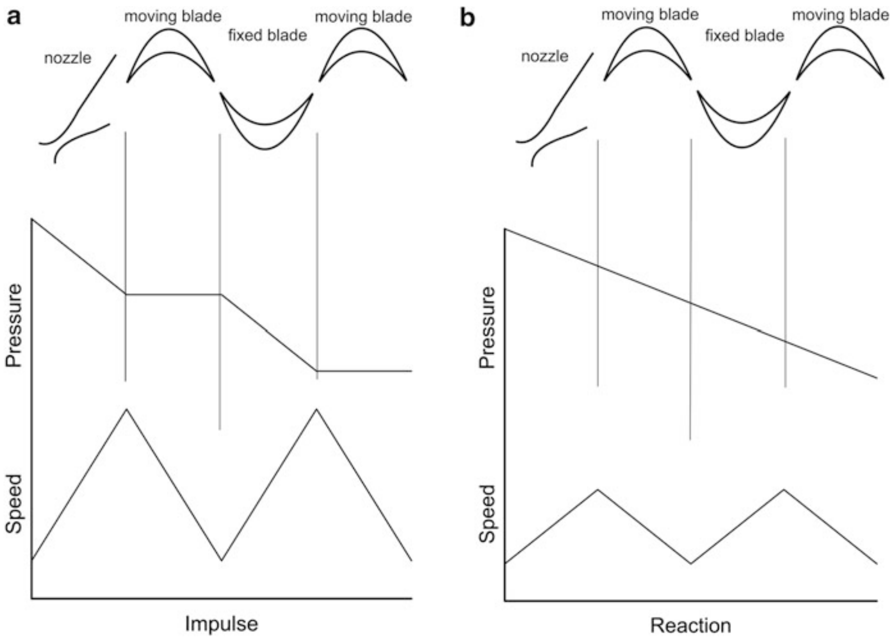


Fig. 11.10 Comparison of the two basic turbine types (**a**) an impulse turbine and (**b**) a reaction turbine

The blades are close together, both rotating and fixed, and the flow passage is a narrow curved slot of sinuous shape.

Each stage of nozzle and blade pair can be designed as either an impulse or reaction type; these have different passage shapes and represent the limiting case, and in practice some turbines are designed with passages representing a mixture of different proportions of impulse and reaction effect. The two types are compared in Fig. 11.10.

In an impulse turbine, Fig. 11.10a, the steam is accelerated in the nozzles by making the pressure fall, according to Bernoulli's theorem, but the blade passage is

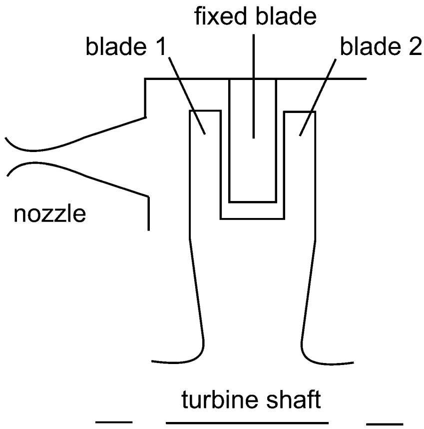


Fig. 11.11 A Curtis velocity-compounded stage

designed so that there is no pressure reduction there. This suggests a uniform cross-sectional area for flow, but to accommodate pressure reduction due to friction, the cross-sectional area is increased slightly. There is a speed reduction in the rotating blade slot, however, because the flow imparts its momentum to the rotating wheel (note, speed not velocity, the flow direction is not being considered here). The next nozzle set accelerates the flow again in exchange for a further pressure reduction, and so on. The other type is the reaction turbine, Fig. 11.10b, in which the slot dimensions are chosen so that the pressure falls uniformly through both nozzles and blades. The speed increases in the nozzles and falls in the blades as before, but speeds are lower than in the impulse turbine because the designer's aim is to make the force on the blade the result of change of momentum plus a force due directly to pressure drop.

That the pressure remains constant through an impulse turbine blade simplifies construction because there is no pressure gradient to cause the steam to leak past the blades around their ends, and labyrinth-like seals are unnecessary. Clearances in the radial direction between moving blades and the casing and between diaphragm and rotor can be relaxed from their usual very small values.

In about 1897, Curtis in the USA adopted what is referred to as the velocity-compounded impulse stage. It is shown diagrammatically in Fig. 11.11 and is usually the first stage of the turbine. The steam leaves the nozzle at a speed twice that shown in Fig. 11.10a. Its momentum is transferred in two increments; half of the speed is lost in the first set of moving blades, and the remaining half in the second set, to bring the steam speed down to a low level again, usually in a manifold. The second manifold is useful in allowing the steam pressure to become circumferentially more uniform as it may have entered the turbine through a partially complete ring of nozzles, a common arrangement for the steam entry. In consequence, both of the Curtis stage blades may be carried on a single disc between entry and exit manifolds—Fig. 11.11.

The first stages of either type of turbine lose efficiency because the blade height is short, so that a large proportion of the surface area in the steam passage is that of the outer ring and the inner rotor surfaces; these provide no contribution to work but contribute to frictional losses. The nozzles turn the steam through an angle without doing work as the nozzles do not move, but at the cost of a further frictional loss. In addition, if the nozzle ring is incomplete, at any particular time only a portion of the blade ring is being supplied with steam and is able to produce work. The friction losses result in heating of the steam and a recovery of some of the specific enthalpy drop, an effect referred to as a "reheat factor" (Wrangham [1948]—in the author's opinion, older textbooks are a good source of information on turbine design; see also Kearton [1945]). The mean circumferential velocity of the blades is a design factor and may be adjusted by the designer in the way the increase in annular area per stage is provided for—an increase is necessary because of the increasing specific volume of the steam as it passes through the machine. The stage increase can be provided by adding length to the blades at their inner or outer radii, or at both ends, or by increasing the root radius.

Finally, it remains to demonstrate what is involved in the design of the steam passages. The problem is one of fluid mechanics, not thermodynamics, and is based on the steady flow energy equation in the Bernoulli form (Sect. 4.2.4). The early days of turbine design predate computers and calculators, and the design equations bear a resemblance to the approach to flow through an orifice plate as set out in Sect. 8.3.1, with factors introduced to allow for frictional losses, etc. An orifice plate in a pipe is, after all, a variable cross-sectional area flow passage. The area of the passage between turbine blades is rectangular and must match that between the nozzles, but each is varied along the flow to produce the speed or pressure change required and curved to produce the change in direction, thus giving a change in velocity that can be analysed with a velocity diagram as shown in Fig. 11.12.

Figure 11.12a, the absolute velocity of the steam leaving the nozzles is v_1. The nozzles are stationary, but the blades are moving to the left, and the relative velocity of v_1 to the blades is u_1. The fluid leaves the blade slot with a velocity relative to the blade surfaces, but this has an absolute velocity v_2 when the movement of the blades is accounted for. The velocity diagram is shown as Fig. 11.12b and the component of most interest is the change in velocity that the steam undergoes through the blade passage, which is Δv. If the mass flow rate is $\dot{m}$, then the force on the blade per blade slot is $\dot{m}\ \Delta v$ (kgm/s^2 or Newton). This total force is at an angle to the direction of motion and the actual force driving the blade is $\dot{m}$(DE), the component normal to the axis of rotation, which produces the torque to drive the alternator. The change in momentum associated with the velocity CE causes the end thrust on the rotor; this velocity component has not been drawn in the figure, for clarity. The blade speed varies with the radius at which the blades are set on the rotor disc, and it is easy to see that the nozzle and blade angles must vary with radius. Very long turbine blades, near the exhaust into the condenser, for example, must be twisted as a result.

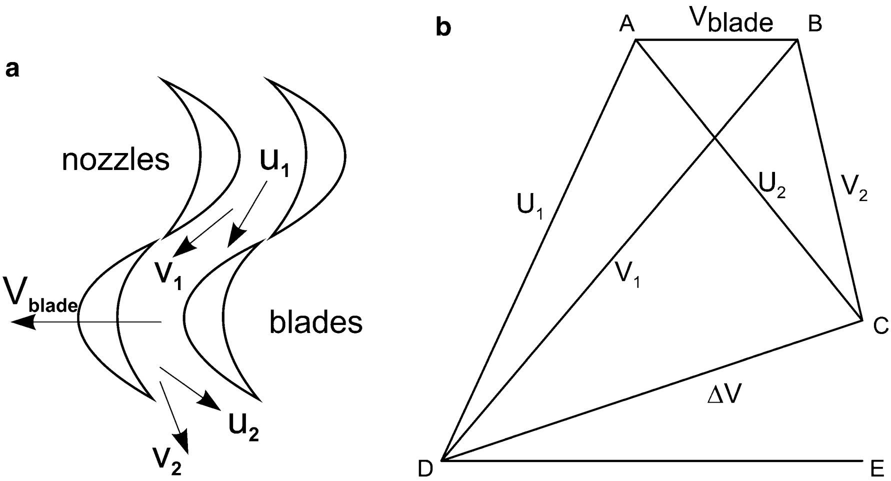


Fig. 11.12 Showing as (**a**) the flow passages in a nozzle and turbine blade pair and the absolute and relative steam velocities, and (**b**) the velocity diagram plotted to arrive at the torque and axial thrust of the pair. The axis of rotation (not shown) is vertical

11.3.3 Transient Performance of Steam Turbines

The response of the turbine to changes in load is important in power stations of any type, and the characteristics are quantified by the turbine manufacturer. In a recent study Kosman and Rosin [2001] report the effect of start-up and load variation on the service lifetime of turbine components. The speed of rotation must be kept constant in order to produce the correct frequency, 50 or 60 Hz, and elaborate speed governors are used. In addition, the power output must balance the load presented to the alternator, so the steam governor must be capable of providing both small and large variations in the flow rate. The required variation is achieved by restricting the flow area through which steam can enter the turbine, thus reducing the pressure, which results in no significant loss of specific enthalpy (throttling takes place) although there is a loss of opportunity to produce work. On machines with a large steam flow, the entry to the nozzles is sometimes through several ports, each with a valve fitted, and the number of valves passing steam is increased as the demand rises, to avoid the difficulty of adjusting a single large valve by a small amount.

Figure 11.13 shows the Willans line, the variation between steam consumption and power output, which was found experimentally by Willans in 1888 to be a linear relationship (Wrangham [1948]) if the turbine is governed by throttling. The line has a slope B and is described by

$$\dot{m}_{turb} = B.(L_e) + \dot{m}_0 \tag{11.1}$$

where

$\dot{m}_{turb}$ is the total steam flow rate to the turbine (kg/s)
L_e is the electrical load (MWe)

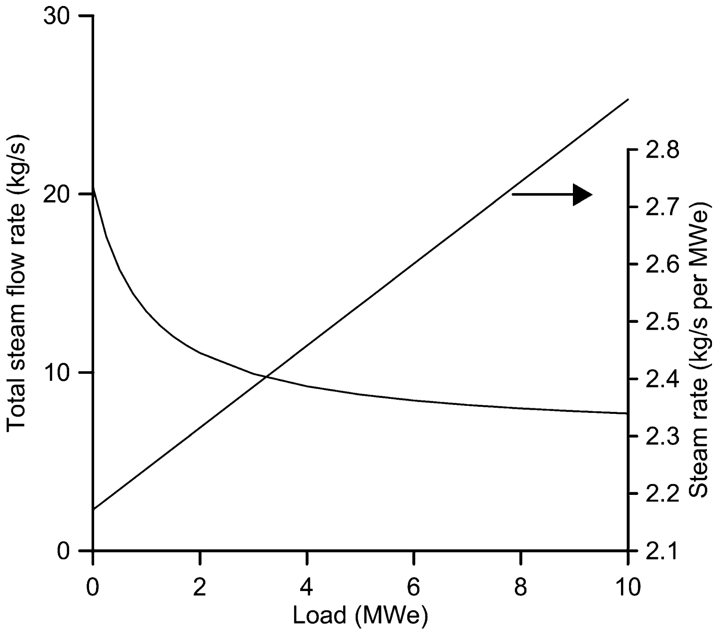


Fig. 11.13 The Willans line for a throttle governed turbine

$\dot{m}_0$ is the minimum steam mass flow rate required to keep the turbine at synchronous speed as load tends to zero (kg/s). Kearton [1945] suggests that $\dot{m}_0$ is 10–14 % of the full load mass flow rate.

An inverse curve is shown, representing $\dot{m}_{turb}/L_e$ in kg/s per MWe, which becomes asymptotic to the Load axis at a value of steam flow per MWe (kg/s per MWe) which is useful to characterise the turbine performance and is sometimes called the steam rate. The rate increases as the load decreases because the steam is throttled and the blades, although rotating at the same speed, are not working at their design condition. For a geothermal condensing steam turbine operating at six bars abs inlet pressure, the steam rate is typically in the range 2.2–2.4 kg/s per MWe.

Geothermal steam is low pressure, high specific volume, which means that the physical size of the governing valves is large for the power output of the turbine as compared to fossil-fuelled plant, and the steam inlet pipe carrying the governor valve is of large diameter. Together with the presence of silica in the steam, this prompted Mitsubishi Heavy Industries Ltd [1998] to develop a butterfly governing valve.

A further issue in relation to transient steam turbine performance is non-base load use. As explained in Chap. 10, their cost structure makes base load use of geothermal power stations very desirable, as availabilities of around 95 % can be achieved. A practical reason making base load use preferable arises because of the very small clearances necessary between the moving and fixed parts of the turbine to restrict steam leakage between stages. Small differential temperatures between

fixed and moving parts are required if the clearances are not to be reduced to zero, when the parts would touch. It is necessary to increase the temperature of the turbine slowly, and manufacturers specify a maximum rate of temperature increase for particular designs. In the past it was the practice in some power stations to pass hot air through the turbine before steam was admitted. Kearton [1945] states that the Ljungstrom radial flow steam turbine warms up much more quickly than an axial flow turbine; they were built in capacities up to 50 MWe and have been examined recently by Marcuccilli and Thiolet [2010] with a view to application to geothermal organic Rankine cycle plant.

11.4 Using Geothermal Steam

11.4.1 Flashing the Well Discharge at Several Pressures

The discharge characteristics vary from well to well, and wells are grouped and their output flashed at a predetermined pressure. There is merit in keeping the turbine inlet pressure and hence the inlet temperature as high as possible, because this increases the conversion efficiency. The higher the flash pressure, however, the more energy is left in the separated water, but it can be flashed at a lower pressure and more steam produced. This could be added to the same turbine fed by the first flash partway through its length, which would then be called a pass-in turbine or mixed pressure turbine—an example has already been shown in Fig. 11.8. In liquid-dominated geothermal resource use, two or three flash stages only are used, as illustrated in Fig. 11.14. However, as will become clear in Chap. 12, the choice of flash pressure and the number of stages depends on geochemical considerations and not just on turbine performance.

Wairakei, which is an old design made complicated by unusual circumstances explained in Chap. 14, has a large number of relatively small turbines. The steam

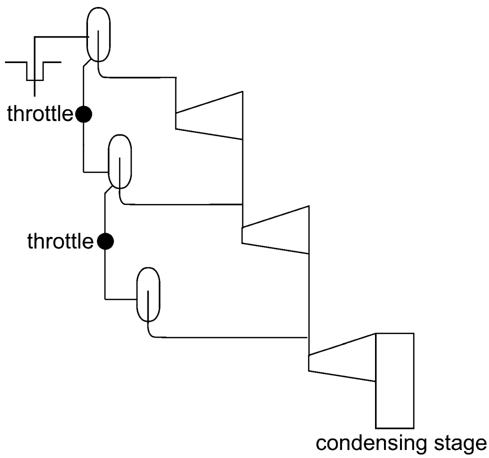


Fig. 11.14 Well discharge flashed in three stages

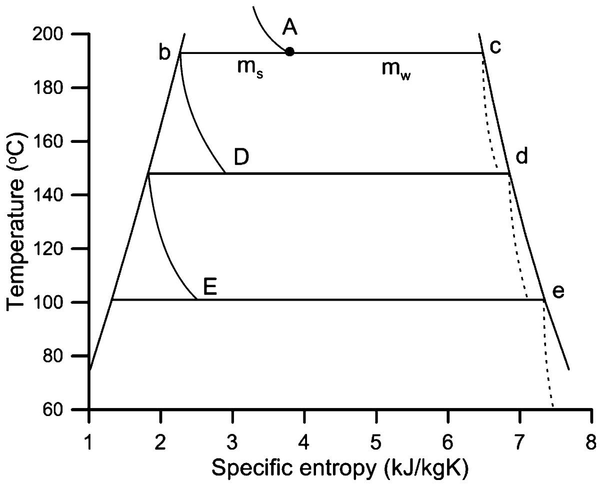


Fig. 11.15 T–s diagram for a triple flash well discharge

was supplied at 13.4, 4.45 and 1.0345 bars abs, with the exhaust of the higher pressure machine combined with new steam flashed from separated water, and the last machine exhausting into a condenser—see Thain and Carey [2009] for details of the arrangement.

The optimum wellhead pressure for a given well discharge characteristic using either single flash or double flash can be calculated, but even adjacent wells may have different characteristics and the choice of type and number of turbine units is the result of a wider optimisation, involving manufacturing costs as well as thermodynamic performance, within the restrictions imposed by resource geochemistry. The theoretical optimum considering only the discharge and flow separation may have a small influence on the adopted design.

The steam in a geothermal power station does not follow a cycle in the plant itself, but the entire combination of resource, well and steamfield equipment and power station can properly be viewed as a cycle. To compare different systems, an analysis based on the second law of thermodynamics must be carried out, usually called "exergy analysis". This is not dealt with in this book, but has been defined by ASTM [2006] and is set out in detail by DiPippo [2012] and other thermodynamics texts. Exergy analysis is increasingly being applied to larger entities than power stations, e.g. country regions (see Sciubba et al. [2008]), and is worthy of a more extended treatment than is possible here.

To calculate the power output for a proposed number of flash stages, a T–s diagram such as that of Fig. 11.15 is required. This figure has been drawn to represent the Wairakei stage pressures mentioned above. Local atmospheric conditions dictate the temperature at which heat is rejected so there is no merit in continuing the T–s diagram to absolute zero. It is usually assumed that specific enthalpy is conserved in a flash expansion, and if the fluid in the resource is a high-temperature liquid, the isenthalpic line enters from above left and ends at the first flash pressure, with saturation temperature of 193 °C, at point A. The mixture is

then separated into saturated water and steam at opposite sides of the T–s envelope, points b and c, respectively. The simple arithmetic rules governing the properties of a two-phase mixture have been explained in Sect. 3.4.2 and already used above to analyse the Carnot cycle. The mass proportion of steam and water are as depicted in the figure, calculated using

$$h = h_f + Xh_{fg} \tag{11.2}$$

where h is the specific enthalpy of the resource fluid and the property values and dryness fraction are for the appropriate temperature, in this case 193 °C. The dryness fraction is

$$X = \frac{\dot{m}_g}{\left(\dot{m}_g + \dot{m}_f\right)} \tag{11.3}$$

so the mass flow rates are proportional to the length of the lines bA and Ac. The separated water from the first flash arrives at point D after the second flash, and likewise at point E after the third. The steam produced is at 148 °C and 101 °C and is represented by points d and e. The exhaust from the high-pressure turbine must be added to the calculated mass flow rate entering at e from the second separated water flash, and the same when considering the final turbine inlet. Since the steam entering any turbine stage is saturated, it will be wet on leaving, and interstage separators may need to be used to ensure that the ongoing steam is dry saturated at inlet to the turbines.

11.4.2 Geothermal Steam Turbines

Geothermal steam carries with it water, dissolved solids and noncondensable gases. The steam supply from a liquid-dominated geothermal resource is likely to be carrying some water containing dissolved silica and calcium carbonate. Solid deposits on turbine blades are the result, and Kubiak and Urquiza-Betran [2002] considered the effect on performance of a reduction in flow area and shape of the passage between blades and nozzles. At some point through a condensing turbine, the steam crosses the envelope of the T–s diagram and becomes wet. The water droplets do not behave like steam in the flow passages, as their momentum is not passed on efficiently to the blades, and they damage the blades and nozzles by their impact. The steam mass flow rate is gradually reduced as the steam becomes wetter. Circumferential water drains are fitted, designed to catch water droplets and remove them from the flow. Non-condensable gases actually produce work in flowing through the turbine, but this and more is required to remove them from the condenser, as discussed below. Turbine maintenance necessitated by the dissolved solids carried in the steam has been addressed by Sakai et al. [2000] and by Matsuda [2006], with discussion of stress corrosion cracking and surface damage by impact and corrosion.

11.4.3 The Possibility of Superheating Using an External Heat Source

Using an external heat source would in principle allow the geothermal steam to be superheated and reheated, similar to Fig. 11.7, which would result in a higher electricity output. Several studies have been carried out; see Kestin et al. [1980]. However geothermal steam carries dissolved solids which could give deposition and corrosion problems in superheaters. Furthermore, there is no obvious source of heat at a high enough temperature that could not be used more effectively in a non-geothermal power station; however, particular circumstances might exist that would make the idea economically attractive.

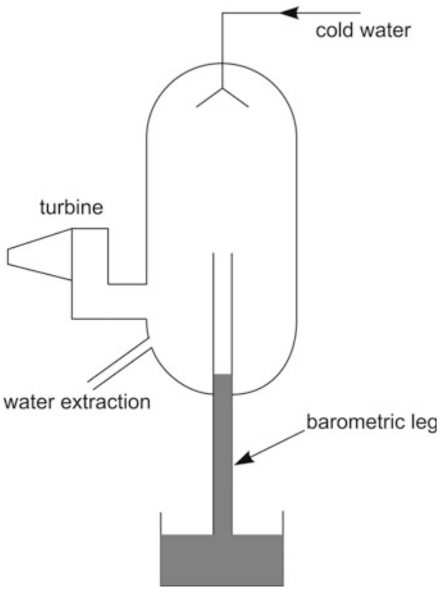
11.4.4 Heat Rejection at Geothermal Steam Power Stations: Condensers and Related Equipment

The problems of heat rejection from geothermal steam plant are greater than those for fossil-fuelled plant in that the geothermal steam carries with it the non-condensable gases mentioned above. However condenser construction is simplified by being able to mix the steam to be condensed with the cooling water because the steam is not part of a high-purity water cycle, allowing direct-contact or jet condensers to be used, in which steam emerging from the turbine enters a vessel into which jets of cold water are sprayed. Once mixed, the cooling water and steam condensate cannot be separated. The alternative is to keep the steam and cooling water separate by using a tubular condenser with cooling water on the inside of a bank or matrix of tubes exposed to steam on the outside. These have a long history of use in fossil plant, but slight differences are required for geothermal plant, such as gaps through the tube matrix to allow the passage of non-condensable gases, a provision which reduces the pumping power required to maintain the vacuum. Tubular condensers are not unknown in geothermal stations. Modes of condensation and non-condensable gas blanketing of condenser surfaces have already been discussed in Sect. 7.5.

Figure 11.16 shows the idea of a barometric direct-contact condenser, named because of its resemblance to a barometer (Fig. 3.1). The vessel contains the spray equipment and sits on a large diameter vertical tube approximately 10 m long, either in a pit or above ground. Imagine a valve on the bottom of the tube, opened when the vessel and tube are filled with water. The vessel would be evacuated as the water ran out and would remain so in use as long as the water level was maintained in the tube and provided the non-condensable gases were continuously removed. The condensate collects in the tube and the water level and hence vacuum are maintained by allowing it to flow out of the tube end. If this is in a pit, the water may have to be pumped out, with further consumption of electricity.

In fossil-fuelled power plant, the turbine is often supported immediately above the condenser, which is physically larger, so that the steam passes through the minimum length of large diameter duct—large diameter because the specific

Fig. 11.16 Diagrammatic arrangement of a barometric leg direct-contact condenser



volume of the steam at low pressure is very large and the flow rate from the turbine exit is slowed to minimise the kinetic energy loss.

Kestin et al.'s [1980] study contains a detailed discussion of condensers for geothermal use, including materials of construction, which is still very relevant.

Although geothermal steam power stations produce more condensate than is required to top up wet cooling towers, the decreasing availability of cheap, clean water for cooling fossil and nuclear power plant has led to a greater use of air-cooled condensers. Very large ones have been built in Europe, including natural draft concrete cooling towers of the configuration usually used for wet cooling. More often finned tubes are arranged in two banks set at an angle, referred to as A frame.

Condenser vacuum is often maintained by extracting the non-condensable gases using steam-jet ejectors in stages. The ejector is a device based on Bernoulli's equation, in which steam, an available, compressible, high-pressure fluid, is accelerated through a nozzle to reach a minimum pressure at its throat—less than the required condenser pressure—after which the flow area is increased, the flow decelerates and the pressure rises. The throat is connected to the condenser where the non-condensable gases collect, and the combined output passes into a small jet condenser, the vacuum for which is produced by a second-stage ejector. The pressure of the fluid leaving the second-stage ejector is above atmospheric so requires no more power consumption to be discharged to atmosphere, although the concentration of H_2S requires it to be discharged at high level or dispersed in some way. The steam consumption of the ejectors is significant, and alternative ways of producing vacuum are used, as reviewed by Ozkan and Gokcen [2010].

Multistage radial flow gas compressors are common, illustrated by Mitsubishi Heavy Industries Ltd [1998], and in the case of high gas-output resources can be physically large items of equipment. Liquid ring vacuum pumps are also in use, the principles of operation of which are shown by one manufacturer, Nash [2012].

The final stage of heat rejection is to transfer the heat collected from the condenser to the environment. Thermal power stations of all types are built adjacent to rivers or estuaries where possible, to reject heat to them (in other words to use them as a source of cooling water). This is the case at Wairakei, where the river water is clean enough to be used in jet condensers; issues relating to water contamination are mentioned in Chap. 14. Alternatively, heat can be rejected to the air by means of cooling towers. On exit from the condenser, the cooling water is sprayed into a rising column of air created by fans on top of an open slatted structure, often with wooden slats; the fans are powered using the generated electricity. The cooled water falls into a pond beneath, from where it is returned to the condenser. The performance of the cooling towers is dependent on the atmospheric temperature. In fossil-fuelled stations natural draft cooling towers are common. The warm condenser cooling water is carried to some height within the tower and sprayed downwards over the cross-sectional area. Atmospheric air enters the bottom of the tower, which is open for a height of only a few metres, unlike the forced draft towers, because the upper part of the tower is shaped like a jet ejector, accelerating the warm air upwards and drawing in fresh air. The towers are made of reinforced concrete. The natural draft cooling tower at Ohaaki, New Zealand, designed for 120 MWe, appears to be the only one of its type used for a geothermal power station. The H_2S extracted from the condenser is released into the rising column of air in the tower, above the water sprays.

11.5 Plant Using Working Fluids Other Than Water

Steam is a suitable working fluid for converting heat into work over the temperature range 160–550 °C at the turbine inlet. Efforts to convert a greater proportion of the energy from fossil fuels led to trials of other working fluids either singly or in combination with steam, but they have found major use only for geothermal generation in the last few decades—in fact they are probably in the majority, numerically. DiPippo [2012] presents a detailed cycle analysis of organic Rankine cycle geothermal plant (and geothermal steam plant) of which there are many in use in different arrangements.

11.5.1 Organic Rankine Cycles

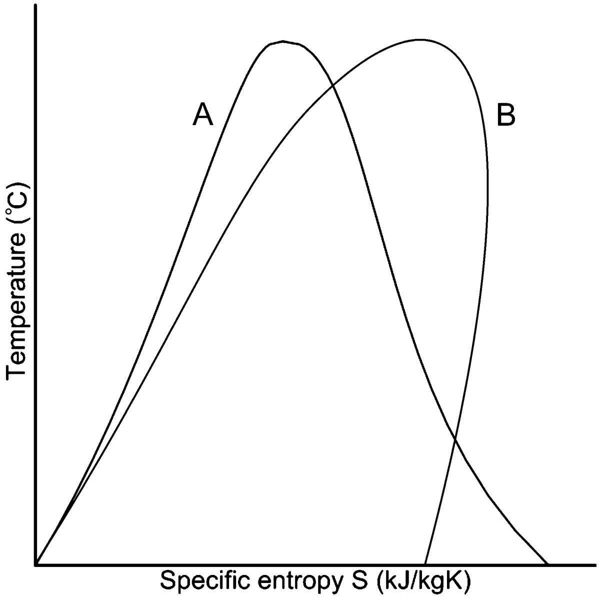
In comparing the Carnot cycle with the fossil-fuelled Rankine cycle in Sect. 11.3.1, the departures of the latter from the ideal were identified. These departures reduced the efficiency of conversion of heat to work and they were to some extent

exacerbated by the shape of the saturation envelope for water. Alternative working fluids were sought at an early stage in power plant development, driven by improvements in the strength of materials at high temperature—Wood [1970] stated that "*claims that some other fluid is preferable to water are a recurring phenomenon*" but went on to note their suitability for low-temperature heat sources. Materials to use high source temperatures were available, temperatures much higher than the critical for water. To retain the advantage of evaporation, a working fluid that was still two-phase at temperatures of 500 °C or more was needed. Rogers and Mayhew [1967] present a numerical example of a "binary cycle", one with two working fluids, water and mercury, operating in two separate but combined cycles with separate turbines. Mercury had some of the required theoretical features but was abandoned in the face of practical difficulties.

Many modern geothermal plants for resources producing water or two-phase well discharges operate with binary cycles, steam coupled with an organic working fluid. The latter necessarily operates in a closed cycle as it is expensive and unsuitable for discharge to the environment. The aim is to convert more of the heat to work than can be achieved using steam plant. Angelino and Colonna di Paliano [2000] identify the same issue in respect of molten carbonate fuel cells—a large proportion of the heat released cannot be used in the primary conversion device. They provide a review of heat recovery to various organic working fluids which are then used with a "conventional" ORC plant. The choice of working fluid for any application is governed by the degree of hazard in its use from the point of view of toxicity, potential for atmospheric pollution, fire hazard, ease of transport, cost and availability, etc. Many different working fluids are currently being considered for use with enhanced geothermal resources; see, for example, Kalra et al. [2012]. In the meantime, using "conventional" geothermal resources, a significant amount of installed geothermal ORC plant is of Ormat Technologies [2012] design and manufacture, using isopentane or *n*-pentane as working fluid. The critical point for isopentane is 187 °C and 33.8 bars abs and geothermal separated water is available at similar temperatures or less.

Although not shown in this book, the h–s chart rather than the T–s chart is the choice of steam turbine designers, and the P–h chart is the choice of ORC designers. For the present purposes, the T–s chart best displays the points to be made. Figure 11.17 shows the general shape of the T–s saturation envelope for isopentane and that for water; several other potential working fluids also have the important characteristic of sloping backwards on the gas side of the envelope, so that s decreases as T decreases. The envelopes of Fig. 11.17 have been superimposed to compare the shapes, and the actual property values are quite different; plotted correctly the two envelopes would be widely separated and of different sizes. Curve A is a proper representation of the T–s envelope for water, but curve B has been drawn by eye and is not meant to represent any particular organic fluid, but to have the general characteristics that have a bearing on its thermodynamic performance as a working fluid.

Fig. 11.17 Comparison of the general shapes of T–s envelopes: (**A**) for water and (**B**) a sketch representing some organic working fluids, including isopentane. The envelopes are not drawn to scale or placed in their correct relative positions on the axes



The important differences in respect of Rankine cycle plant design that follow from the envelope shapes are as follows:

- An isentropic expansion of saturated vapour results in a wet mixture at turbine exit for H_2O and a superheated vapour for working fluid B, provided that the expansion starts at a suitable temperature. This removes concern about blade erosion by droplets.
- The slope of the liquid side of the envelope is less steep for B than for water so the departure from the Carnot cycle is greater for the organic fluid, which, taken in isolation, would tend to make regenerative heating more attractive than for steam Rankine cycles.
- The saturation envelope for B is fairly wide near its summit, which is beneficial in maximising the proportion of heat added during evaporation and helping to reduce the effect of departure from the Carnot cycle resulting from heating in the liquid state (see Fig. 11.2).
- At 30 °C, a suitable temperature for rejection of heat to the environment and thus the lowest temperature potentially required in the cycle, the saturation pressure of isopentane, in particular, is approximately 1 bar abs. This is mechanically convenient as the condenser leakage potential is minimised.
- The density of gaseous organic fluid is high, which leads to a comparatively smaller turbine for a given output—the rate of transfer of momentum in the blades increases with density.
- Choosing an upper temperature of 220 °C for separated geothermal water, and a heat rejection temperature of 30 °C, the specific volume ratio for saturated

vapour, V_{220}/V_{30}, is 382 for water and approximately an order of magnitude less for isopentane and other organic fluids. As for density, this too permits many fewer stages for an ORC turbine than for a steam turbine working between the same temperatures (recall Parsons' original reasoning in Sect. 11.3.2).

- The velocity of sound in organic fluids is lower than in steam, and turbine blade tip speeds must be kept subsonic. This might require a gearbox between turbine and alternator, to increase the alternator revs/min to synchronous speed.

The current range of plant using isopentane typically has a two-stage turbine (Legmann and Sullivan [2003]) in line with the observations above. As regards regenerative feed heating, the small number of stages in the turbine reduces the opportunity to bleed working fluid—the flow is large enough but the temperature differentiation is probably too small and some heat would be transferred using larger than necessary temperature differences, thus defeating the plan to approach the Carnot cycle requirements. However the advantages of regenerative heating were its effect in reducing the condenser capacity as well as improving cycle efficiency, and there could be benefit from this aspect. A large proportion of the heat supply is rejected in the condenser because of the low source temperature, which makes the condenser heat load large. The typically air-cooled condensers consume electricity and are physically large. Purely in terms of cycle efficiency, regenerative feed heating would be more beneficial to organic working fluid cycles than to steam cycles in view of the slope of the T–s envelope on the liquid side. A low-temperature heat source is available in the separated geothermal water flow.

As regards superheating or reheating, the slope of the T–s envelope on the gas side makes it unnecessary to adopt superheating or reheating for the reasons it was adopted in steam cycles. The maximum temperature of the heating fluid must be taken advantage of, and deviation from the Carnot cycle is not a reason for avoiding superheat, which many organic Rankine cycles employ in their optimised configuration. Sohel et al. [2010] report on a dynamic model of the performance of a 5.4 MWe organic Rankine cycle power plant of unspecified design. They provide a T–s diagram showing that the working fluid is superheated, but leaves the turbine before reaching condenser conditions, entering a heat exchanger (termed a recuperator) which transfers heat back into the working fluid heating part of the cycle, perhaps exchanging work from the turbine for a reduced condenser heat load.

There is a great deal of current literature concerned with selecting a working fluid from the large range of organic fluids available. For example, Franco and Villani [2009] provide a detailed discussion of the methodology and factors for design optimisation of this type of plant for geothermal fluid temperatures in the range 110–160 °C. Such is the range of fluids on offer that their focus is on methodology rather than engineering factors.

Perhaps the most noticeable features of the majority of ORC plant, of Ormat Inc design and manufacture, are that they can stand in the open, and the prime mover is small, very much smaller in bulk than the heat exchangers which are shell and tube type. The condensers are air cooled with the working fluid contained in almost horizontal banks of finned tube with cooling fans—the tubes are set at an angle to

the horizontal to allow the liquid to drain for collection. The condensers cover a significant area, but can be elevated above the rest of the plant. These features will have an effect on the engineering–economic optimisation of the design, which thus cannot be analysed from a purely engineering performance standpoint.

Throughout the development history of the fossil-fuelled Rankine cycle reviewed above, the search for optimum economic performance by manipulating the cycle appears to have been the primary focus. It must be remembered however that the full commercial activity of designing and supplying the main power station plant involves research and development, design and then manufacturing which includes consideration of the capital assets of foundry and machine shop, then manpower, finance, transport to site, maintenance, reliability and overall environmental impact and energy analysis of the type discussed in Sect. 10.7—no small task. Thermal efficiency of the plant is important but is by no means the dominant factor in making a selection and cannot be viewed in isolation. In common with all commercially developed power industry equipment, what is marketed is the result of private manufacturing industry optimisation.

11.5.2 An Example of Dual Working Fluid Plant

Legmann and Sullivan [2003] describe the Rotokawa 1 development, New Zealand, which is an example of using steam from the wells directly in a back-pressure steam turbine and using its exhaust together with the separated water to supply organic Rankine cycle generators. The station was designed and manufactured by Ormat Inc. The total designed output of the plant is 30 MWe of which 14 MWe is produced by a multistage reaction steam turbine and the balance by three 2-stage organic Rankine cycle (ORC) turbines. The steam turbine rotates at 3,000 rpm and the ORC machines at 1,500 rpm. A reconstructed process flow diagram from details in the paper is shown as Fig. 11.18, from which it can be seen that two of the ORC units receive their heat supply from the steam turbine exhaust.

The other two operate as a pair in tandem, using heat from the separated water. The process is at its simplest, and the paper reports that the power station is very reliable with an availability of 98 % and output 10 % higher than the design figure.

Ormat supplied a second-stage power station at Rotokawa, this time with a simpler process flow arrangement, shown by Sohel et al. [2009]. The steam flow drives a 35 MWe steam turbine which exhausts to a steam to isopentane heat exchanger, thus supplying the heat for a 7.5 MWe ORC unit. The separated water provides the heat source for a second ORC unit of the same output. Sohel et al. demonstrated that an improvement in efficiency had been achieved for summer operation by increasing the heat rejection capacity in the air-cooled condenser by using water. The gain in power output on the hottest day amounted to 6.8 % but averaged over the year was only 1 %; it is a reflection of the variation of weather conditions and provides no basis for judging the optimum cycle configuration adopted by the plant designers.

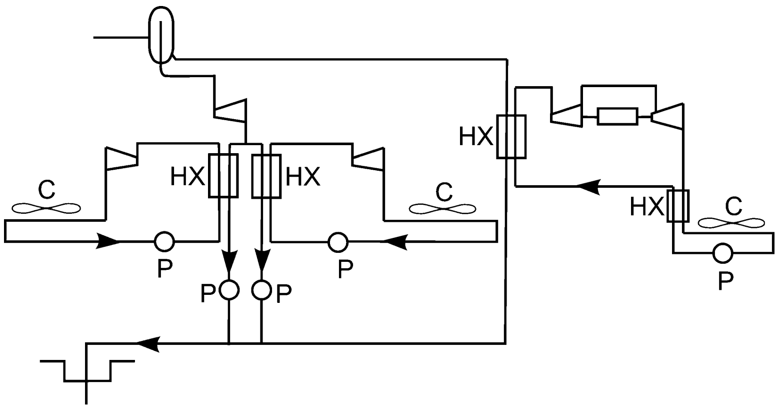


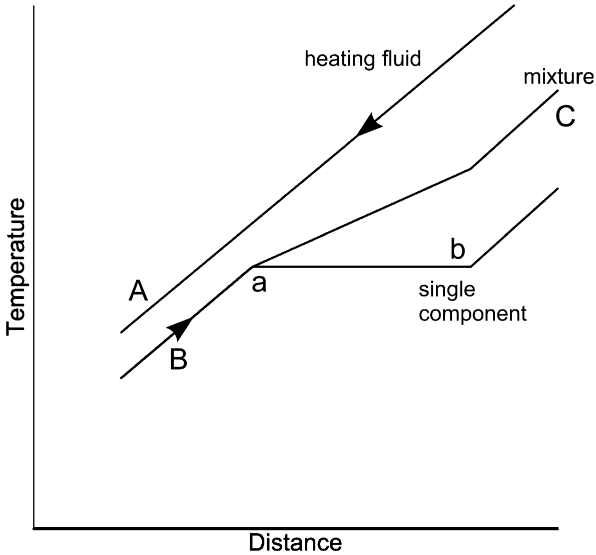
Fig. 11.18 Interpretation of the combined steam and ORC plant described by Legmann and Sullivan [2003]. The closed loops contain isopentane, HX indicates heat exchangers, P pumps and C air-cooled condensers

11.5.3 The Kalina Cycle

The Kalina cycle first seems to have entered the geothermal literature via the paper by Kalina and Liebowitz [1989] which considered the application of a cycle using an ammonia–water mixture as working fluid.

It will be recalled from Sect. 3.3 that Carnot's ideal circumstances for converting heat to work required that any temperature reduction must produce work; otherwise, the heat flow would be wasted. It was convenient for the Carnot cycle that evaporating liquid remained at constant temperature, because it provided the ideal constant temperature heat source. Now consider transferring the heat from a geothermal liquid, say separated water, in a long heat exchanger. The heating liquid remains single phase and its temperature gradually falls between inlet and exit as it transfers its heat to another fluid. Figure 11.19 represents the situation, with the separated water heating fluid entering from the right and experiencing falling temperature as it transfers its heat, line A; consider temporarily that the horizontal axis represents the distance through the heat exchanger. Travelling in the opposite direction is the working fluid of an ORC, entering from the left at lower temperature, line B. The working fluid heats up to saturation temperature at "a" on the graph and remains isothermal until it is totally evaporated at "b", after which it shows an increase of temperature with position as before. If heat can be transferred at a suitable rate at point "a", then beyond that, the temperature difference is unnecessarily large, and the temperature of the heating fluid is being reduced wastefully. To clarify the details, the horizontal axis represents the amount of heat transferred, which is linearly related to heating fluid temperature change. This graph occurs in heat exchangers from many industries, chemical processing and nuclear, fossil and geothermal power plant, including ORC plant. The benefit of the Kalina cycle is that the working fluid is an ammonia–water mixture, and as the mixture increases in

Fig. 11.19 The fundamental idea behind the Kalina cycle is to reduce the change of temperature difference between the heating and evaporating fluid



temperature, the ammonia comes out of solution and the saturation temperature of the mixture rises. It can be made to vary as the line marked “mixture”, line C on Fig. 11.15, which on average shows a lower temperature drop between the heating fluid and the evaporating mixture. A heat engine which must take its heat supply from a flowing stream of single-phase liquid (i.e. almost constant specific heat) will have a higher efficiency if this line can be followed instead of the single component line.

Potentially, the Kalina cycle has a higher efficiency than the ORC cycle, and a 1997 study by Lu (reported as Lu et al. [2009]) showed that the proposed Kalina cycle design referred to as KCS11 was expected to have a higher efficiency than an existing ORC plant. Welch and Boyle [2010] report on the design and construction of two Kalina cycle plants. DiPippo [2004] carried out a comparative performance analysis of several ORC plants and the only operating geothermal Kalina cycle plant and concluded that the Kalina plant may produce a 3 % higher output for the same operating conditions. The plants examined had different local conditions; the Kalina plant is in Iceland where heat rejection is easier than for the ORC plant in the USA because ambient temperatures are lower. The Kalina cycle plant is more complicated than ORC plant and has not attracted significant capital investment yet. Its proper place in the hierarchy is yet to be established.

11.5.4 The Trilateral Flash Cycle and Two-Phase Prime Movers

The trilateral flash cycle is shown on the T–s diagram of Fig. 11.20, which is a modification of the Rankine and Carnot cycle diagrams of Fig. 11.2 for water as the working fluid; the cycle is proposed for organic working fluids also. Saturated water

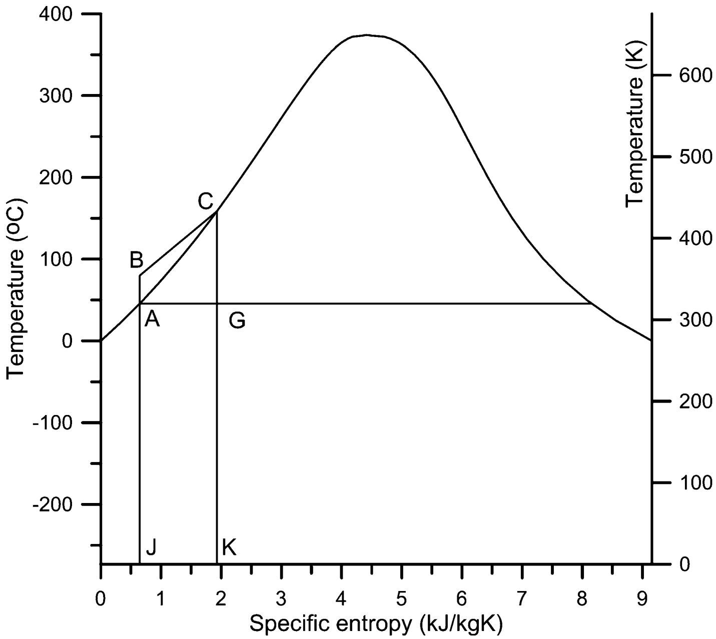


Fig. 11.20 The trilateral flash cycle, plotted on a water T–s envelope

arrives at the condenser at A, is pumped up to maximum pressure at B from where it is heated in the compressed liquid phase to C. From C the water is expanded to G from where it is condensed back to A. The cycle has a triangular shape. Isentropic expansion CG has been shown in the figure, but in reality the fluid state will be slightly right of G as specific entropy increases due to frictional heating in the expanding fluid. This increase in specific entropy may be greater than that in a steam turbine, as the working fluid is saturated liquid at C and thereafter is a disorganised two-phase mixture. The figure has been drawn for a maximum pressure of 6 bars abs and a condenser pressure of 0.1 bar abs, thus working between 159 and 46 °C, and the Carnot efficiency is the ratio of the areas ABCG to JBCK—compare the equivalent areas for the Carnot and Rankine cycles of Fig. 11.2, which uses the same temperatures. The average temperature at which heat is transferred to the fluid is less than in Fig. 11.2, so the theoretical efficiency must be less than that of the Rankine cycle, but the equipment is much simpler.

The cycle as shown here is described as a wet vapour cycle. Smith [1993] notes that Ruths [1924] took out a patent based on his idea that heat stored in an accumulator could be used to advantage by discharging the whole amount through a reciprocating steam engine or turbine. The geothermal equivalent of Ruths' proposal is to discharge a well producing two-phase flow directly into a prime mover. Smith et al. [1995] provide a review of suitable prime movers developed for the geothermal and chemical industries, including the bi-phase turbine and screw expander, and he later focused on geothermal use—Smith et al. [2005]. The screw

expander has continued to attract attention but few have been built; an early one built in the USA has recently been replaced by a standard ORC plant (Buchanan and Nickerson [2011]) but that is barely significant given the small amount of development effort applied to their mechanical design. Welch and Boyle [2010] provide information on a recently developed type.

The fluid in a two-phase mixture does not travel at a uniform speed and, intuitively, should give a lower conversion efficiency than the well-ordered flow through a turbine. However Brown and Mines [1998] compared an ORC plant and a trilateral flash plant and concluded that the prime mover for a trilateral flash cycle system would require a machine efficiency of only 76 % to give it the same overall conversion efficiency of heat to work as an ORC plant with a typical turbine of 85 % mechanical efficiency. As before, the thermodynamic cycle issues are easy to address in considering the use of the trilateral flash cycle and two-phase prime movers, and the real problem is a lifetime energy (exergy) optimisation—is more thermodynamic work expended in manufacturing the equipment than is generated by it, and are the total environmental effects worthwhile? Smith et al. [2005] presented reasons for incorporating screw expanders into ORC plants, providing evidence of increased output and decreased cost.

References

Angelino G and Colonna di Paliano P (2000) Organic Rankine cycles (ORC) for energy recovery from molten carbonate fuel cells AIAA 2000-3052, 35th Intersociety Energy Conversion Engineering, Nevada, USA

ASTM American Society for Testing and Materials (2006) Standard guide for specifying thermal performance of geothermal power plants, ASTM E974-00

Brown BW, Mines GL (1998) Flowsheet simulation of the tri-lateral cycle. Geoth Resour Counc Trans 22:105

Buchanan T and Nickerson, L (2011) Expansion and repowering of mammoth geothermal resource: selection of generating cycle and expander technology. Geothermal Resources Council Annual Meeting, San Diego, USA

DiPippo R (2004) Second law assessment of geothermal binary plants generating power from low temperature geothermal fluids. Geothermics 33:565

DiPippo R (2012) Geothermal power plants: principles, applications, case studies and environmental impact. Butterworth-Heineman, Oxford

Franco A, Villani M (2009) Optimal design of binary cycle power plants for water dominated medium temperature geothermal fields. Geothermics 38(4):379–391

IAEA (2012) World energy supply, Bulletin 46/1 http://www.iaea.org

Kalina AI, Liebowitz HM (1989) Application of the Kalina cycle technology to geothermal power generation. Geother Resour Counc Trans 13:605

Kalra C, Bequin G, Jackson J, Laursen AL, Chen H, Myers K, Hardy HK, Zia J (2012) High potential working fluids and cycle concepts for next generation binary organic Rankine cycles for enhanced geothermal systems. In: Proceedings of 37th workshop on geothermal reservoir engineering, Stanford, USA

Kearton WJ (1945) Steam turbine theory and practice. Pitman, London

Kestin J, DiPippo R, Khalifa HE, Ryley DJ (eds) (1980) Sourcebook on the production of electricity from geothermal energy. US DoE, Brown University, Rhode Island

Kosman G, Rosin A (2001) The influence of startups and cyclic loads of steam turbines conducted according to European standards on the component's life. Energy 26:1083

Kubiak JA, Urquiza-Betran G (2002) Simulation of the effect of scale deposition on a geothermal turbine. Geothermics 31:545–562

Legmann H, Sullivan P (2003) The 30MWe Rotokawa I geothermal power project; 5 years of operation. Iceland Geothermal Conference

Lu X, Watson A, and Deans J (2009) Analysis of the thermodynamic performance of Kalina cycle system 11 (KCS11) for geothermal power plant—comparison with Kawerau Ormat binary plant. In: Proceedings of ASME Energy Sustainability Conference, San Francisco, USA

Marcuccilli F, Thiolet D (2010) Optimising binary cycles thanks to radial inflow turbines. In: Proceedings of World Geothermal Conference, Bali, Indonesia

Matsuda H (2006) Maintenance for reliable geothermal turbines. GRC Transact 30:755

Mitsubishi Heavy Industries Ltd (1998) Geothermal power generation, a company booklet, revised edition

Mitsubishi Heavy Industries Ltd (2012) http://www.mhi.co.jp

Nash (2012) http://www.gdnash.com/e-library.aspx#howitworks_

Ormat Technologies Inc (2012) http://www.ormat.com

Ozkan NY, Gokcen G (2010) Performance analysis of single flash geothermal power plants: gas removal system point of view. World Geothermal Congress

Rogers GFC, Mayhew YR (1967) Engineering thermodynamics, work and heat transfer. Longman, London

Ruths J (1924) Method and means of discharging heat storage chambers containing hot liquid used in steam power and heating plants. UK Patent 217,952

Sakai Y, Yamashita M, Sakata M (2000) Geothermal steam turbines with high efficiency, high reliability blades. GRC Transact 24:521–526

Sciubba E, Bastianoni S, Tiezzi E (2008) Exergy and extended exergy accounting of very large complex systems with an application to the province of Siena, Italy. J Environ Manage 86:372

Smith IK (1993) Development of the trilateral flash cycle system part 1: fundamental considerations. Proc Inst Mech Eng 207:179–194

Smith IK, Stosic N, Aldis C (1995) Trilateral flash cycle system – a high efficiency power plant for liquid resources. WGC, China

Smith IK, Stosic N, Kavacevic A (2005) Screw expanders increase output and decrease the cost of geothermal binary power systems. Geother Resour Counc Trans 29:787–794

Sohel MI, Sellier M, Brackney LJ, Krumdieck S (2009) Efficiency improvement for geothermal power generation to meet summer peak demand. Energy Pol 37:3370

Sohel MI, Krumdieck S, Sellier M, Brackney LJ (2010) Dynamic modeling of an organic Rankine cycle unit of a geothermal power plant. In: Proceedings of World Geothermal Congress

Thain IA, Carey B (2009) 50 years of power generation at Wairakei. Geothermics 38(1):407–413

van Riemsdijk JT, Brown K (1980) The pictorial history of steam power. Octopus Books, London

Welch P, Boyle P, Sells M, Murillo I (2010) Performance of new turbines for geothermal power plants. GRC Transact 34:1091–1096

Wood B (1970) Alternative fluids for power generation. Proc I Mech E 184(1):1969–1970

Worley NG (1963) Steam cycles for advanced Magnox gas-cooled nuclear power reactors. Proc Inst Mech Eng 178(Part 1, No 22):1963–1964

Wrangham DA (1948) The theory and practice of heat engines. Cambridge University Press, Cambridge